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DYNAMIC ANALYSIS OF MULTIBODY SYSTEMS FOR RECIPROCATING COMPRESSOR OF HOUSEHOLD REFRIGERATOR

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ABSTRACT

In spite of development of compressor technology in its capacity, there are still many problems to improve with respect to noise and vibration. In order to understand the mechanism of the noise generation and reduce the vibration and noise level, it is necessary to understand the dynamic characteristics of compressors. In this study, suspension spring is modeled as flexible body, and the other parts of reciprocal compressor are modeled as rigid bodies. Each rigid body and spring is connected with joints. The dynamic analysis is performed with DADS. Using the compressor model, the characteristics of frame vibration is examined as the variation of offset, connecting rod, and eccentricity, by comparing the change of side pressure and frame behavior. With these results, weight balance is redesigned. The vibration characteristics of the frame is compared with experimental results.

1. INTRODUCTION

The performance of a compressor has much effect on the performance of refrigerators. In spite of development of compressor technology, there are still many problems to improve. Especially, vibration and noise characteristics of compressors are important criteria to determine the quality level of refrigerator. In order to understand the mechanism of the noise generation and reduce the vibration and noise level, it is necessary to understand the dynamic characteristics of compressors.

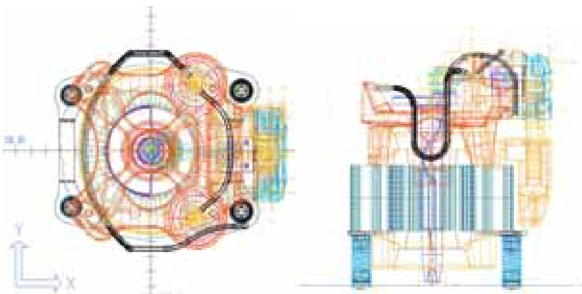
Reciprocating compressors is widely used, because they have a simple mechanism and good efficiency, and they are not sensitive to the gas condition in suction and compression process. In spite of the simple structure, the mechanism of vibration generation is complicated. There are many sources of vibration and noise in the structure of a compressor (Lee, 2003). Because of its small size, it is difficult to analyze its dynamic characteristics.

The vibration and noise of compressors are generated from motion of mechanism of compressor sustained elastically. This motion is created by motion of piston and connecting rod, and motion of valve, force created by unbalance of rotating parts, the impulse created by magnetic force, and so forth.

In this study, the reduction of vibration and noise is examined through dynamic analysis of mechanism of compressor. It is assumed that mechanism of compressor is rigid and the spring is modeled as flexible body, and the dynamic analysis is performed with DADS. The change of side pressure of piston is examined with changes of offset, length of connecting rod, and eccentricity. Weight balances are redesigned in order to reduce unbalance force for rotating force of connecting rod and inertial force of piston. The vibration of frame is analyzed and compared with measurement of frame acceleration.

2. COMPRESSOR MODEL

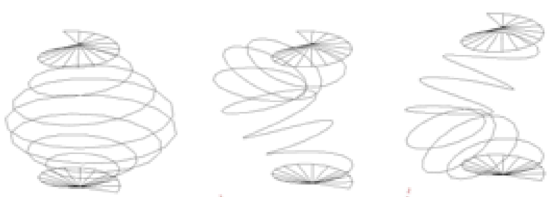
The compressor model consists of shaft, connecting rod, piston, piston pin, frame, and suspension spring. The coordinate system for the model is defined as Figure 1. Reference frame is set at the intersection point of the centerline of shaft and the plane constituting bottoms of 4 springs. Z coordinate is the centerline of shaft, and X direction corresponds to the direction of motion of piston.



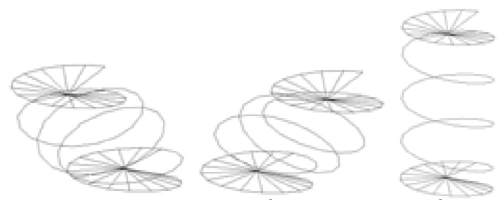
(a) X-Y plane view (b) Y-Z plane view
Figure 1: Global coordinates of compressor

Table 1: Type of Joints

Joint	Joint members
Revolute	Frame – Shaft
	Piston pin – Piston
Cylindrical	Shaft – Connecting rod
Spherical	Connecting rod – Piston pin
Translational	Piston – Frame



(a) 1st mode (b)2nd mode (c)3rd mode
Figure 2: Normal modes



(a) 1st mode (b)2nd mode (c)3rd mode
Figure 3: Static constraint modes

Each part of mechanism is assumed rigid, and rigid bodies are connected by joint. Types of joints connecting rigid bodies are shown in Table 1. RSDA (Rotational Spring Damper Actuator) element is used to model frame-shaft, and equivalent torque data is applied as input according to rotating speed of shaft. Also TSDA (Translational Spring Damper Actuator) element is used in piston-frame, and pressure data is used according to rotation angles of shaft. The compressor model is operated by these two elements.

There are four springs to sustain the frame of compressor, and they are modeled as flexible body that has 6 degrees of freedom (x, y, z, ϕ_x, ϕ_y, ϕ_z). FEM models of springs are created with CBEAM of NASTRAN, because it is easy to model, and simple to post-process, it can save calculation time (Ahn, 1998). The spring model has 4 beam elements in each turn. The length of moving part excluding the part fixed at boss is considered as effective length of the spring. Each spring is mounted on frame and ground by spherical joint and bracket joint. To describe dynamic behavior of spring, its natural mode must be determined by analysis of flexible body.

When flexible body moves, if it is constrained at its both ends or exposed to load, the deformation mode is hard to obtain accurately by natural mode. In that case, static mode must be determined (Kim, 2001). There are static attached mode and static constraint mode in static mode. Generally, static attached mode is used for flexible body with large inertial force and static constraint mode for the body with small inertial force such as spring. Lanczos method is used to extract natural mode in the modal analysis.

To determine the natural mode of spring, both ends of spring is fixed on upper and lower bosses, and stiffness matrix, mass matrix, eigenvalue, and eigenvector are calculated. To determine constraint mode of spring, unit displacement is given to each joint to get results. We give unit displacement at the joint of frame and joint in X, Y, and Z direction. At this time, all mobility is constrained except mobility in the direction of unit displacement. In this study, normal modes are calculated up to 10th mode, and static constraint modes up to 3rd mode. Normal mode shapes are shown in Figure 2, and static constraint mode shapes in Figure 3.

The DADS software provides spring and bushing elements for modeling a spring. Since the spring element has 1 degree of freedom, it is not enough to describe dynamic characteristics of the spring. On the other hand, the bushing element has 6 degrees of freedom. So, bush model of the spring is also constructed with the bushing elements. For flexible body model and bush model of the spring, the behavior of frame is compared. The results of flexible body model and bush model show consistency in dynamic behavior, but the flexible body model shows a more realistic behavior.

From these results, it is not easy to judge which one is more appropriate to describe dynamic behavior of frame. The bush model needs stiffness and damping coefficient in every 6 directions to model with. It is difficult, however, to obtain precise data for stiffness and damping coefficient. The flexible body model needs only elastic coefficient, density, Poison's ratio of material to analyze, and it is preferable to use the flexible body model. Figure 5 shows complete configuration of the DADS model. Names with suffix –flex mean spring data files, and names with prefix cy-, re-, and sph- mean joints between rigid bodies.

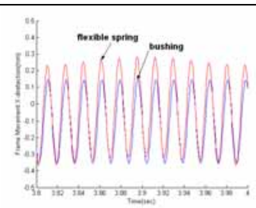
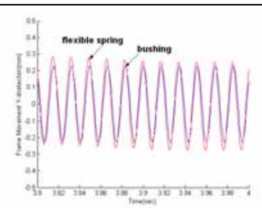
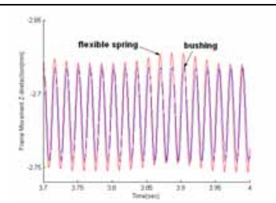
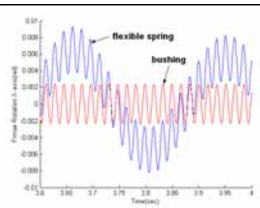
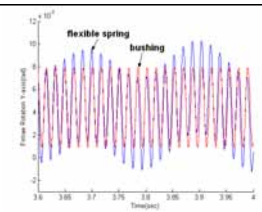
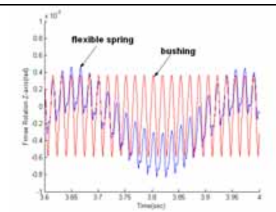
	X-direction	Y-direction	Z-direction
Frame Movement			
Frame Rotation			

Figure 4: Frame behaviors of flexible body model and bush model for the spring

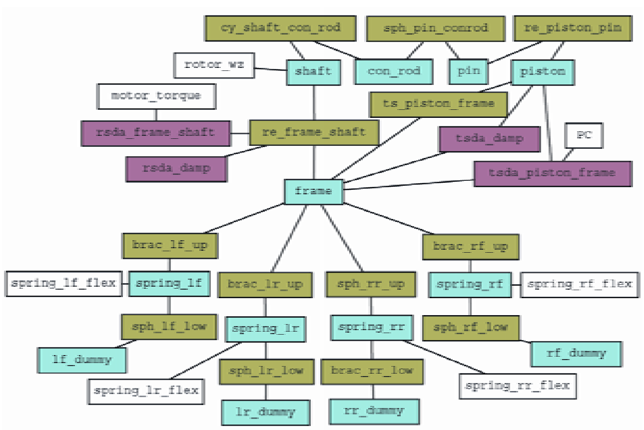


Figure 5: DADS modeling map

3. SIMULATION

3.1 Inputs and Initial Conditions

For rotation of shaft, initial angular velocity is given, and torque is applied to RSDA element of frame-shaft according to rotation speed of shaft. As shown in the torque curve of motor in Figure 6, combined wire torque data are used until the maximum speed of 2540rpm. After that, torque data is switched to the primary wire torque (Ryff, 1994). As internal pressure of cylinder, TSDA element of piston-frame uses experimental data shown in Figure 7. The data are in absolute pressure. The deflection of frame due to weight is simulated. The deflection reaches 2.77mm in 2 seconds as shown in Figure 8.

Transient behavior of compressor is shown in Figure 9 from 0 rpm of initial condition to steady state. Rotation speed of shaft approaches to steady state in 0.22 seconds. In this model, the rotational velocity of steady state is 3515 rpm, and shows a fluctuation of $\pm$53.7rpm.

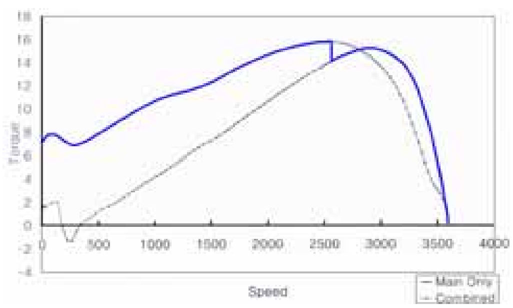


Figure 6: Motor torque curve

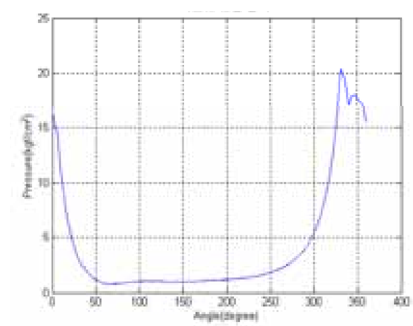


Figure 7: Pressure versus rotor angle

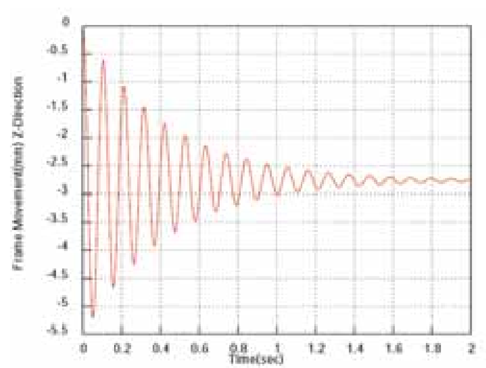


Figure 8: Z-direction movement of frame

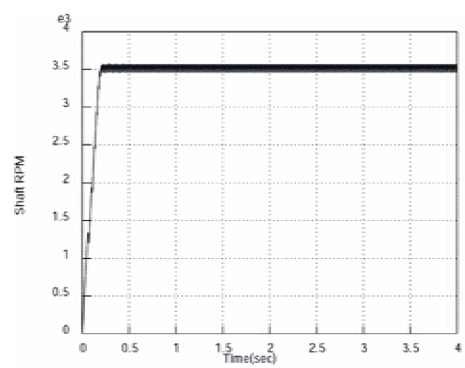


Figure 9: Simulation results of shaft rotational velocity

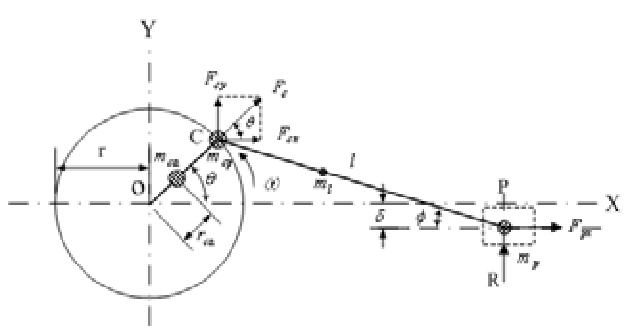


Figure 10: Forces acting on compressor

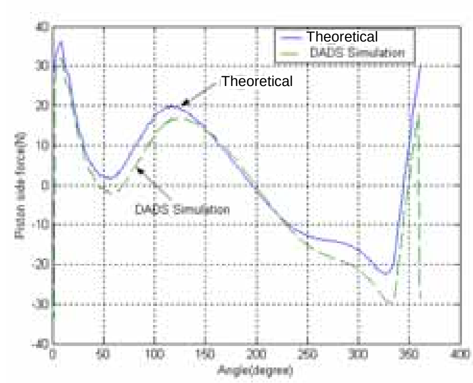


Figure 11: Side force

3.2 Side Pressure

The side pressure of piston acts on cylinder wall when the rotation motion of crank changes to linear motion, and is the main cause of mechanism vibration. As shown in Figure 10, there are force P acting in the direction of center axis of piston by pressure of piston, and the inertial force F_{px} that is given in Equation (1).

$$F_{px} = (m_p + m_{lr}) r\omega^2 \left(\cos\theta + \frac{1}{\lambda}\cos 2\theta - \frac{\delta}{l}\sin\theta \right) \tag{1}$$

The resultant force $F = P - F_{px}$ acting on center axis of piston pin gives side pressure R of Equation (2) on the wall of cylinder.

$$R = F \cdot \tan\phi = \pm\left(\frac{\pi \cdot D^2}{4} p - F_{px} \right) \sqrt{\frac{\left(\frac{1}{\lambda}\sin\theta + \frac{\delta}{l} \right)^2}{1 - \left(\frac{1}{\lambda}\sin\theta + \frac{\delta}{l} \right)^2}} \tag{2}$$

where $\lambda = \dfrac{l}{r}$, ϕ is the angle of connecting rod, and m_{lr} is the equivalent reciprocating mass of connecting rod. Figure 11 shows good agreement for theoretical analysis and simulation results.

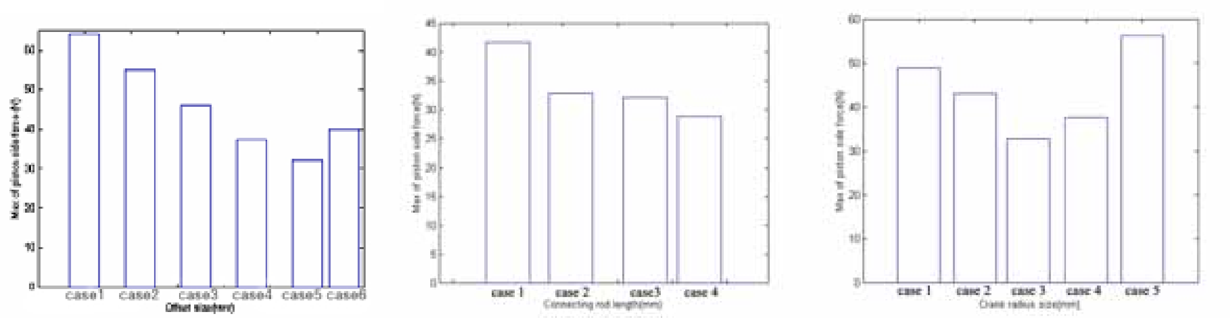
4. PARAMETRIC ANALYSIS

4.1 Offset

The offset is defined as distance from the rotational center of shaft to the center axis of piston in y-direction. The size of offset is increased by 0.6mm from zero, and the results for 6 cases are compared and analyzed. The simulation results in Figure 12(a) show that side pressure is highest in case 1, and decreases according to increase of offset. It is lowest in case 5 of 2.4mm offset, and increases in case 6 again.

4.2 Connecting Rod

The length l of connecting rod is the one from the center of crank pin to the piston pin. As the ratio $k = l/s$ (s means stroke) increases, the side pressure of piston acting on the cylinder wall decreases, the mechanical efficiency increases, the torque curve becomes flat, and inertial force increases. In Figure 12(b), the length of connecting rod is 40mm in case 1, 42.3mm in case 2, 45mm in case 3, 47mm in case 4. It shows side pressure of each case. It is observed that the longer the length of connecting rod is, the lower the side pressure is.



(a) offset (b) connecting rod length (c) rotation radius of crank

Figure 12: Maximum side force with respect to the change of parameters

4.3 Eccentricity

The eccentricity is the rotational radius of crank, which is a length from rotational center of shaft to the center of crank pin. Since the stroke is determined by the eccentricity, as the eccentricity becomes small, the ratio $k = l/s$ increases, and the side pressure to wall of cylinder decreases. The effect of eccentricity is examined by changing the eccentricity size from 7mm to11mm by the step of 1mm. Since the size of eccentricity affects not only stroke but also volume of cylinder, the radius of pistons is adjusted to keep the same exhaustion. In the simulation, the same plot of pressure is used for internal pressure of cylinder. Figure 12(c) shows that the side pressure decreases and then increases as the eccentricity increases. It is observed that case 3 of 9mm eccentricity shows lowest side pressure.

5. REDUCTION OF VIBRATION

5.1 Redesign of weight balance

The relationship of piston-crank mechanism in the compressor is shown in Figure 13. The unbalance force is generated by the centrifugal force F_c of crank and the reciprocating inertial force F_{px} during the rotation of crank. This unbalance force is the main cause of vibration in compressor. The weight balance is the one of methods to reduce this kind of unbalance force. In this study, the effect of weight balance is examined by comparing the acceleration at the frame for basic model and redesigned model with simulations.

The weight balance is attached to the opposite side of crank pin, and reduces rotational unbalance moment of crank. The mass of weight balance must be determined by examining reciprocating inertial force equilibrium as well as rotational force. Inertial force F_{px} is described as Equation (3)

$$F_{px} = (m_p + m_{lr})r\omega^2\left(\cos\theta - \frac{\delta}{l}\sin\theta\right) + \frac{1}{\lambda}(m_p + m_{lr})r\omega^2\cos 2\theta \qquad (3)$$

As shown in Equation (3) Inertial force is sum of the 1st inertial force and the 2nd one. To remove the 1st inertial force, weight balance $(m_p + m_{lr})$ is attached to the opposite side of crank. It can eliminates the unbalance force of X-direction, however, the unbalance force of Y-direction $(m_p + m_{lr})r\omega^2 \sin\omega t$ is generated. Therefore, half balancing method is used to cancel the centrifugal force by reciprocating inertial force by adding weight balance of $(m_p + m_{lr})/2$. The mass M_b and rotational radius r_b can be determined as Equation (4).

$$M_b r_b = \left[m_{lc} + m_{cp} + m_{ca}\left(\frac{r_{ca}}{r}\right) + \frac{1}{2}(m_p + m_{lr})\right]r \qquad (4)$$

where m_{lc} is the rotational mass of connecting rod.

Considering the crank mechanism of Figure 13, the weight balance is designed separately and added with the crank arm as shown in Figure 14. To avoid structural interference in redesign of weight balance, its thickness t_b is limited to less than 7mm, inner and outer radii r_{bi}, r_{bo} are fixed to the same size of the basic model. The crank mechanism is redesigned as follows: the mass M_b of weight balance is 0.0673kg, radius r_b is 11.87mm, center angle 2α is 208°, and thickness t_b is 7mm.

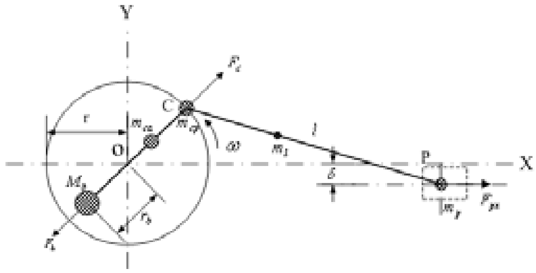


Figure13: Crank mechanism

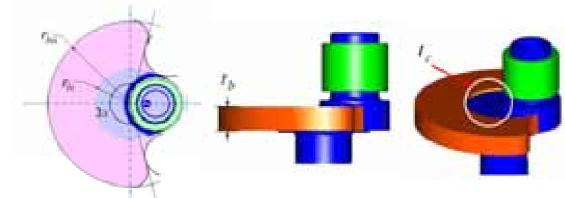
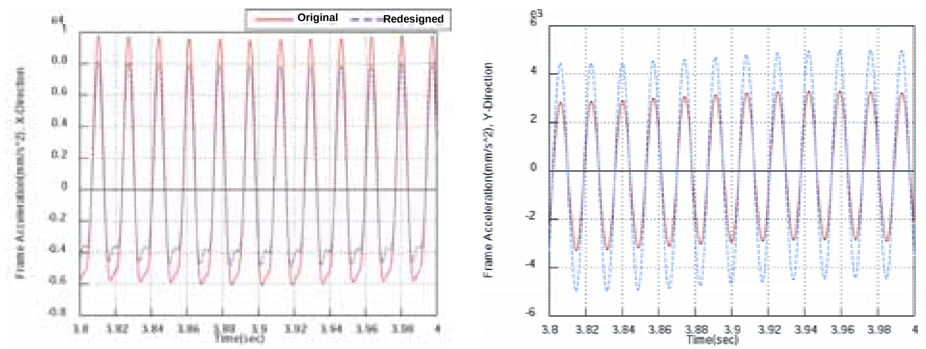


Figure 14: Weight balance



(a) X direction (b) Y direction

Figure 15: Acceleration at C.G. of frame

In the original design of weight balance, compressor operates in unstable state due to relative unbalance of X and Y-direction. The redesigned weight balance reduces unbalance of X-direction and increases unbalance of Y-direction a little bit, so that operation of compressor becomes more stable. From the result of simulation of Figure 15, accelerations at the mass center of frame are about -6~10m/s^2 in X-direction, and -3~3m/s^2 in Y-direction for original model. In redesigned model, we get -5~8m/s^2 in X-direction, and -5~5m/s^2 in Y-direction.

5.2 Measurement of vibration

To examine the effect of weight balance, vibration of frame in the compressor is measured, and compared with the results of simulation. The configuration of measurement system is shown in Figure 16. A 3-axis accelerometer is installed on the stator of compressor to measure vibration of frame. The accelerometer is installed far from valve position to avoid direct affection of valve opening and block deformation during operation of compressor. The accelerometer is B&K Type4326, and to avoid current noise of stator, insulating material is used in attachment of accelerometer.

PID controlled pressure regulator is used to regulate internal pressure of compressor. After inlet and outlet pressures of compressor reached the condition of Ashrae, the vibration of frame is measured. As defined in Ashrae condition, outlet pressure is adjusted to 13.9kg$_f$/cm^2, inlet pressure to 1.176kg$_f$/cm^2.

The acceleration is measured from the original model and the redesigned model of compressor. To examine improvement effect of weight balance, frequency component of 58Hz (nominal rotational speed 3515rpm) for X-direction acceleration is compared.

As shown in Figure 17, the acceleration in X direction is 0.552G in original model, and 0.276G in redesigned model. This shows a reduction of 50%. The acceleration of Y-direction is also reduced from 0.406G to 0.385G, and it also shows a favorable effect on the vibration of Y-direction. Unbalance due to X and Y-direction acceleration in the redesigned model is also improved as compared to the original model.

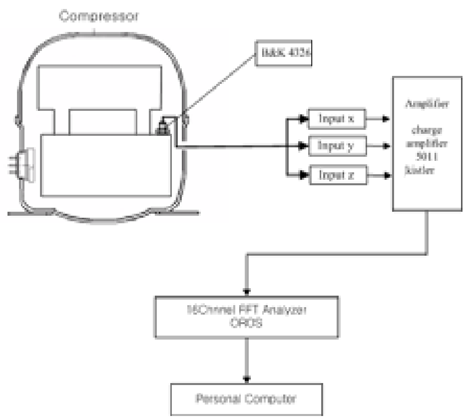


Figure 16: Experimental setup

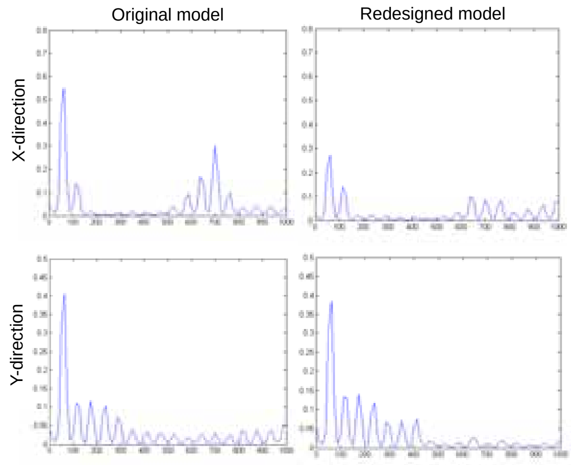


Figure 17: Auto-spectrum of frame acceleration

6. CONCLUSION

To reduce vibration and noise of a compressor, its dynamic characteristics is examined and the effect on vibration and noise is analyzed. Mechanism of the compressor is modeled as rigid bodies. The spring is modeled as a flexible body. Using the compressor model, the characteristics of frame vibration is examined as the variation of offset, connecting rod, and eccentricity, by comparing the change of side pressure and frame behavior. With these results, weight balance is redesigned. The frame vibration characteristics of the redesigned model is examined by experiment.

From the result of side pressure, as offset increases, the side pressure increases and then decreases to attain its lowest value at the offset of 2.4mm, and then increases again. In case of connecting rod, the longer its length is, the smaller the side pressure is. As size of eccentricity increases, the side pressure decreases and then increases again, and the side pressure has lowest value in eccentricity of 9mm.

From the result of simulation and measurement for weight balance, the redesigned model shows reduced unbalance force of frame in X and Y-directions. Acceleration is also reduced, which means an improvement of vibration of compressor. It can be concluded that the vibration of compressor can be improved by changing the parameters of compressor mechanism. It is considered that these results can be a guide in the design of compressor to reduce vibration.

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